ZAMCON CORPORATION (A)*

Design of a Steam Turbine Bearing Cover

In May 1965 Jud Noble, a Senior Engineer with the Propulsion Division of the Zamcon Corporation in Galveston, Texas, found that two bearing covers in a steam turbine for a ship propulsion unit were not strong enough to meet the contract specifications. Results of the stress analysis on the bearing covers had been delayed pending completion of a computer program needed to find the loads on the covers. Meanwhile the turbines were being built. When Jud found that the two covers were over-stressed, there did not seem to be time to design and cast new, stronger covers without throwing the project seriously behind schedule.

In September 1964 Zamcon had received contracts to build the propulsion units for 26 small surface ships being built in the United States for a friendly foreign power. The propulsion units each consisted of two steam turbines of about 18,000 hp -- high pressure and low pressure, to be used in cross-compound operation -- plus the reduction gearing and the condenser. Irving Ashby, Design Engineer, was responsible for the overall design of the two turbines, which were to be built in accordance with a new set of shock design specifications for calculating the shock loads transmitted to various parts of the ship by underwater explosions which occur near the vessel but do not rupture the hull. It is particularly important that all parts of the propulsion equipment be strong enough to survive the severest expected shocks, otherwise the ship will be disabled. This would be the first time Zamcon had designed equipment to the new shock specifications.

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* Fictional name. All names of firms, places, government agencies and officials, and Zamcon employees which appear in this case have been disguised.

Prepared in the Design Division of the Mechanical Engineering Department, Stanford University, by John A. Alic under the direction of H. O. Fuchs, with financial support from the National Science Foundation.

Zamcon is one of the nation's largest corporations, a diversified manufacturing enterprise with annual sales of many millions of dollars. The Propulsion Division manufactures propulsion equipment both for government agencies and commercial ships. This includes the steam turbines and reduction gearing, condensers and air ejectors, controls, and turbine-generator sets. Although most of the Galveston operations are devoted to marine equipment, the well-equipped shops are occasionally used for the manufacture of such things as stationary steam equipment, and the launch equipment for missile carrying submarines.

Irving Ashby and Jud Noble both work in the Marine Engineering Department; there are about 130 people in the department, 50 of whom are engineers. As a Design Engineer, Irving oversees an entire project. Much of his work is non-technical, involving rather complex dealings with the government, as well as with shipyards and with other departments at Zamcon. Irving's own specialty is the thermodynamics of the turbines -- efficiences, steam and water rates, pressures, temperatures, etc. He is able to utilize the services of specialists in other fields, such as mechanical design, manufacturing methods, and materials and metallurgy when designing turbines.

A Zamcon negotiations engineer working with the Sales Department secured the orders for the propulsion units. The government agency acting as a purchasing agent for the vessels was the Department of Vessels and Engines (DOVE). After DOVE had sent the specifications for the ships to many shipyards, about a dozen submitted bid proposals. Even before this Zamcon had sent each of the shipyards a proposal for the propulsion units so that the yards could incorporate this information in their bids. Competitive propulsion unit manufacturers did the same. Four shipyards, one of which was designated the "prime contractor", were awarded contracts by DOVE for the 26 ships; Zamcon then received contracts from the shipyards for propulsion units.

The vessels were to be built over a period of several years at a cost of almost $11 million each. The firm of Ahab & Sons, Inc., prominent Seaport City marine engineers and consultants, had designed the vessels for DOVE. Ahab & Sons also acted as the "design agent" for the prime shipyard; Zamcon dealt mostly with them rather than directly with DOVE or the prime contractor.

The ships are 450 feet long. They carry guns, guided missiles and depth charges. The equipment within the vessel must be strong enough to resist damage caused by shock from the explosion of the vessel's own depth charges, as well as from underwater concussions as a result of enemy action.

When Zamcon received the first contract for propulsion units, Irving Ashby was sent a copy. While an existing design was to be used for the low pressure (LP) turbine, a new high pressure (HP) turbine had to be designed. Outline drawings showing the relationships between the com-

ponents of the propulsion unit appear in Exhibit 1. These drawings dimension all connections between the propulsion unit and the ship and between the individual parts of the propulsion unit. Irving explained that it would ordinarily take 16 to 18 months to build such a unit on a government contract. This is for the first unit, the design and construction of which are scrutinized most heavily by the design agent and the government agency involved. However, propulsion units for commercial ships can usually be completed in about a year because the relations between the supplier and the customer are so much more flexible. Requirements which government equipment must meet are contained in various general specifications; these are modified for specific classes of ships. The process of winning approval of design and construction methods and quality often becomes time consuming. As a further complication, the individual shipyards also have a degree of autonomy in specifying the design.

The design of the HP turbine, begun with the receipt of the contracts, is illustrative of the handling of a new design, although the turbine utilizes no new technology and is similar to others Zamcon builds. Irving designed a turbine in sketch form to expand the steam as desired. From the sketches, designers and draftsmen prepared engineering drawings. At this time much effort went into designing the rotor so that slots for the blades in all stages would align and could be broached simultaneously. These drawings were given to an analytical design group and the critical speeds of the rotor were computed. The first critical speed was found to be within the operating range of the turbine, near one of the usual cruising speeds of the ship. It was then decided to increase the length of the rotor (between the bearings) by 4 inches to lower the first critical speed to a point midway between two of the incremental running speeds. The extra two inches at each end of the rotor provided more space between the gland seals and the bearing oil seal, a welcome side effect.

The LP turbine was originally designed in 1953; some 20 to 30 had been built previously. An assembly drawing of the turbine appears in Exhibit 2. Items 62 and 63 comprise the thrust bearing. Item 2 is the rotor bearing at the thrust end, item 3 that at the coupling end. Each bearing consists of a split ring lined with tin-base babbitt which fits around the rotor journal. The outer surface of the ring is spherical and fits in a spherical cavity formed by the bearing support, item 48 -- a casting fabricated into the cylinder base assembly -- and the bearing cover. The two parts are split horizontally at the rotor centerline. Item 4 is the bearing cover (and end cover) on the thrust end. Item 47 is the coupling end bearing cover. Items 58 are the gland seals, items 54 and 55 the seal retainers. The bearings are self-aligning and are prevented from rotating by a pin fixed to the lower half of the bearing ring. This pin fits into a clearance hole in the bearing support; the hole is actually a slot (to allow for thermal growth) which is closed by the bearing cover. Contact between the bearing ring and its housing is through three "spherical keys" as shown in Exhibit 3. See also

Exhibit 2. The keys are blocks with their outer surface spherical -- they bolt to the bearing ring and their heights can be adjusted with shims. This configuration greatly simplifies the manufacturing of the bearings, for the entire outer surface of the bearing ring does not have to be fitted to its housing; it is just rough machined. Only the spherical surfaces of the keys are hand scraped during fitting. The bearing metal is centrifugally cast inside the bearing ring and relieved in the center of the upper halves of the bearings; Irving explained that turbines are designed so that under no operating conditions does the load vector at the bearing deviate by more than about 45° from vertically downwards. He said that the load is not always straight down because under some operating conditions the steam forces create vectors 90° ahead of the direction of rotation which combine with the static weight vector of the rotor to result in loads off the vertical centerline.

When Zamcon secured the contracts, three engineers began immediately to apply the new dynamic shock analysis method to the propulsion unit. The first components analyzed were to be the bearing covers on the LP turbine, which must restrain the rotor. However the management of the Propulsion Division knew that it would be several months before even the preliminary results of this analysis would be available. The method was complex and a computer program would have to be prepared to apply it. Because the production schedule on the first propulsion unit was tight, and of critical importance to the success of the entire project, management decided to go ahead and begin making the first unit before the results of the shock analysis were available, even though the design might have to be changed when it was already partially completed.

A ship propulsion turbine typically takes 5000 to 20,000 man hours to build, depending upon its size and features. Although some of the smaller parts and subassemblies for the 26 propulsion units were to be made in batches, the larger subassemblies and the turbines themselves would be processed individually, on a staggered schedule, because their size and complexity made batch production impossible. Hopefully, the shock analysis would be completed before more than a few propulsion units were actually under construction.

In September Irving ordered the long lead time castings and forgings for the first LP turbine. Zamcon purchases all castings and forgings and it would take several months to procure them. Irving also sent drawings and parts lists of the LP turbine to the Manufacturing Information section of Zamcon's Industrial Engineering Department. From the drawings, Manufacturing Information prepared method sheets which listed the steps to be followed by the shop personnel in the manufacturing of the turbines. Project scheduling was the responsibility of the Planning and Scheduling Department, while the Purchasing Department had to order purchased parts, coordinating their purchases with the production schedule. Irving had to inform purchasing about the castings and forgings he had already ordered; then, in the coming months he had to keep all concerned parties informed of design changes made to the turbine. He was also responsible for cost control. A PERT chart for the LP turbine appears in Exhibit 4.

On October 30, 1964, Irving sent outlines and assembly drawings of both the LP turbine and the HP turbine plus an outline drawing of the entire propulsion unit (parts of which appeared in Exhibit 1) to Ahab & Sons for approval. Ahab & Sons returned the drawings in December, with requests for various changes. Zamcon acquiesced to some of the proposed changes; however, others became matters for negotiation and would be finally settled only by DOVE. Changes to the LP turbine which Irving had to consider involved such things as adding pipe plugs in several of the castings, changing the type of sight flow indicator (used to check the oil flow in bearings) from a simple glass tube cast into the housings to a more expensive attached gauge, adding a flange to the coupling-end bearing drain hole for a pipe connection, and changing the location of the sentinel valve (which blows if the condenser pressure reaches 5 psig) so that it would be more easily visible. Changes to the LP turbine design were completed and the revised drawings sent to Ahab & Sons by February 3, 1965. After inspecting them, Ahab & Sons sent the drawings, together with a new list of comments and recommendations, to DOVE's Resident Overseer in Seaport City. Zamcon also received a copy of the comments Ahab & Sons had made. After scrutinizing the drawings himself, the Seaport City Resident Overseer sent them on to the DOVE office in Washington, D.C. During this time the Seaport City Resident Overseer was also in touch with the four shipyards through their respective Resident Overseers. All these people had to be satisfied with the design. After being approved by the DOVE office, the drawings progressed backwards through the Seaport City Resident Overseer to Ahab & Sons, who, on May 23rd, issued a design release for the LP turbine to Zamcon. Meanwhile, work on the first propulsion unit was well along; castings and forgings had begun to arrive from suppliers in December and January, and had been placed into process as soon as possible.

In actuality the design of the LP turbine was not completely frozen with the issue of the release. Because the HP turbine was entirely new, there were more questions concerning it, and on June 16th a design review -- a meeting of representatives from Zamcon, the shipyards and DOVE -- was held in Washington at the request of DOVE. At this meeting some final changes to the LP turbine were also decided upon. The design releases eventually secured for all parts of the propulsion unit were still implicitly subject to their meeting the shock criteria of the ongoing dynamic analysis, however.

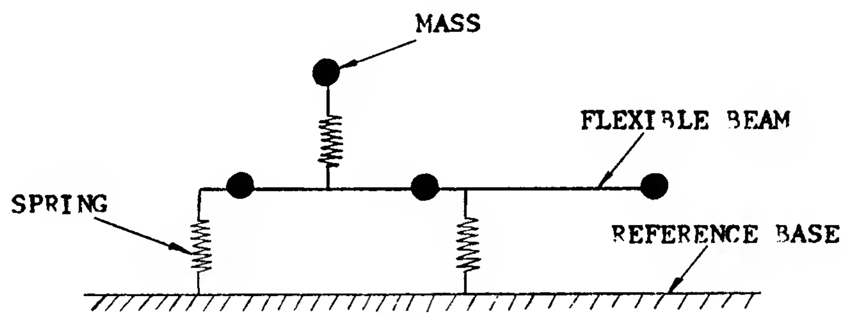
Dynamic Shock Analysis Method

The contracts called for shock analysis in accord with a report written by R. O. Belsheim and G. J. O'Hara* entitled "Shock Design of Shipboard Equipment: Dynamic Design Analysis Method." Excerpts from the Abstract of this paper appear below:

*Real names

A design-analysis method is presented for the calculation of the shock response of . . . shipboard equipments which are not installed on non-linear shock or vibration mounts. It is essentially a simplified modal analysis method where the design shock inputs have been empirically obtained from realistic underwater explosion tests, as reinforced by information obtained from theoretical and laboratory studies. The method allows variations in the complexity of analysis dependent upon the importance classification of the equipment . . . . Differing energy-absorption capacities for different geometries and loadings, when some plasticity can be allowed, is permitted by the use of an effective yield stress as the failure criterion.

The method outlined in the paper employs normal mode vibration theory. The equipment and the part of the ship's structure which supports it are modeled as a system of lumped masses, vertical springs and flexible horizontal beams. A simple two-dimensional model consisting of four masses, three springs and one beam appears below:



The masses are assumed constrained to move only vertically. The springs and beams are considered massless and linear. No damping is included.

When this system is subjected to a shock input at its base -- a suddenly applied displacement or velocity -- it will vibrate. The system has as many degrees of freedom, hence as many natural frequencies and modes of motion, as it has masses. For each natural frequency, or mode, considered individually, there are different relative displacements of all the masses -- hence for each mode there is a "mode shape" relating these displacements. For most systems, in the first, or lowest frequency mode, all the masses move in phase, all reaching their maximum displacements in the same direction at the same time. In any mode all masses reach their maximum displacements, regardless of direction of motion, at the same time. In modes higher than the first some of the masses will always be moving in opposite directions. In calculating the force resulting in the equipment for a shock input, the assumption is made that the modes act independently. Then the resulting loads or stresses in the piece of equipment are combined. If the resultant stresses for all modes were combined the sum would be excessively conservative; thus, DOVE

specifies that only a certain fraction of the total number of modes of the system be considered. Although the model is only an approximation of the equipment, with different natural frequencies and mode shapes, Jud Noble pointed out that stresses and strains calculated from the model are similar to those in the equipment because the shock inputs are based on test results.

DOVE does not specify how a piece of equipment should be modeled, although the entire shock analysis program is subject to their approval. The shock inputs are prescribed, however; these are usually different for each class of ship and are contained in the ship specifications. They are presented in the form of a shock spectrum with step velocity inputs as a function of natural frequency and effective weight. The effective weight depends upon actual equipment weight, mode shape, and natural frequency. A piece of equipment has an effective weight and natural frequency for each mode; thus there is a different shock input for each mode. The shock spectra are derived from experiments.

The limiting part of the analysis is the determination of the various modes and their natural frequencies. A matrix iteration technique is required, hence the need for a computer program in the analysis. Jud said that it was important to pick a model for the system which would represent the actual equipment well while not being impossible to analyze. His problem when modeling the turbine and its supporting structure was how to represent an extremely complex structure including many auxiliary plates, stiffeners and gussets in addition to the main structural elements by a highly simplified system of point masses, springs and flexible beams. Depending on the assumptions he chose to make he had wide latitude in the numbers and locations of each of these elements.

LP Turbine Bearing Cover Stress Analysis

The bearing covers for the thrust end and the coupling end of the LP turbine as originally designed in 1953 are shown in Exhibit 5 and 6. They are steel castings.

During October 1964 a very much simplified shock load analysis was performed to allow a preliminary check of the stresses in the bearing covers. It was recognized from the first that the bearing covers were among the most critical components from a strength standpoint because they must restrain the 11,730 lb. rotor. As a first guess at the shock loads which the complete computer applied shock analysis would give, the LP turbine was modeled as a single mass with velocity inputs applied through a single spring by Dale Evans, an engineer employed by Zamcon in Norfolk, Virginia. During these months the Propulsion Division was in the process of moving from Norfolk to Galveston. Dale found shock loads on the covers in the vertical, athwartships and fore and aft directions by applying the new dynamic analysis method to a single spring and mass to get a "g" value. Knowing the weight of a component, he could then determine a shock load.

Dale found the vertical load on the bearing covers to be the greatest, and he computed stresses in the covers and in the studs which hold them to the bearing supports. He treated the cover as a "thick" half-circular ring and assumed that the load was applied at a single point in the center of the cover by the spherical key. His preliminary analysis is summarized below:

Part	Shock Load (lbs)	Equivalent g's	Stress (psi)	Stress Type	Allowable Stress** (psi)
Coupling End					
--Bearing Cover	147,000	25	36,000	Bending	36,000
--Studs	147,000	25	60,000	Direct	81,000
Thrust End					
--Bearing Cover	147,000	25	Not checked since stronger than coupling end.		
--Studs	147,000	25	40,200	Direct	40,000

These results seemed to indicate that the bearing covers might be acceptably strong. When the bearing cover castings for the first two turbines were received they were put into process. Photographs of the castings appear in Exhibits 7 and 8.

Dale Evans had remained in Norfolk and Jud Noble assumed the task of heading a shock analysis group at Galveston to develop the complete mathematical model of the propulsion unit, along with computer routines for analysing the model. This work began in September 1964 with one engineer working on the model and two working on the computer program used to find the mode shapes. In later stages of the program, while stresses were being determined, one or two more engineers were added. The computer program was made general enough so that it could be used for many different pieces of equipment. The model arrived at for shock in the vertical direction is shown in Exhibit 9. Other models had to be found for shocks in the fore and aft and athwartships directions. While Jud was able to use two-dimensional models for the LP turbine, the HP turbine model had to be three-dimensional for a good representation. Since the condenser hangs from the LP turbine it had to be included in the model. The springs representing its foundations include in their rates the flexibilities of the structure between the turbine casing and the hull. One beam with five masses represents a bundle of 4000 condenser tubes.

** Allowable stress is taken to be equal to the yield stress since a factor of safety is included in the velocity input to the model. In his later calculations Jud Noble used the yield stress specified by DOVE for this material, 30,000 psi. It is likely that the 36,000 psi Mr. Evans used is a supplier or commercial value.

An example of the problems Jud encountered when constructing the LP turbine model was the determination of the combined spring constant for the bearing support, the bearing and its oil film -- the elements interposed between the turbine casing and the rotor. The bearing supports can be broken down into linearly elastic elements and their spring constants determined. Likewise the bearing is an elastic metal ring. The flexibilities of these and other such elements were calculated and the results checked against a considerable backlog of experience and test data available in the engineering department. The oil film between the bearing and the shaft is highly non-linear. Since the model cannot accommodate non-linear springs, a linear spring rate had to be assumed. In order to make an intelligent assumption, Jud had first to complete enough of the shock analysis to get a good estimate of the load on the oil film. He then found that under shock loads the film was so stiff that it could be neglected.

The number and location of the point masses chosen for each component also required careful thought. For a component with little flexibility, such as the turbine casing, Jud felt that it was sufficient to use masses sized and placed to give a total mass, center of gravity and moment of inertia equal to those of the component. For flexible elements where forces would be more greatly magnified, such as the bundle of condenser tubes, he used a greater number of masses.

A model report describing the equipment, the model and the assumptions made in its preparation was submitted to DOVE for approval on February 5, 1965. DOVE requested changes to the model. In the ensuing months work on the model continued and revised model reports went to DOVE in March and in October. Not until May 1965 was the model in acceptable form for design decisions to be based on results from it. The computer program was also not developed to the point where four-beam models could be handled until shortly before this time.

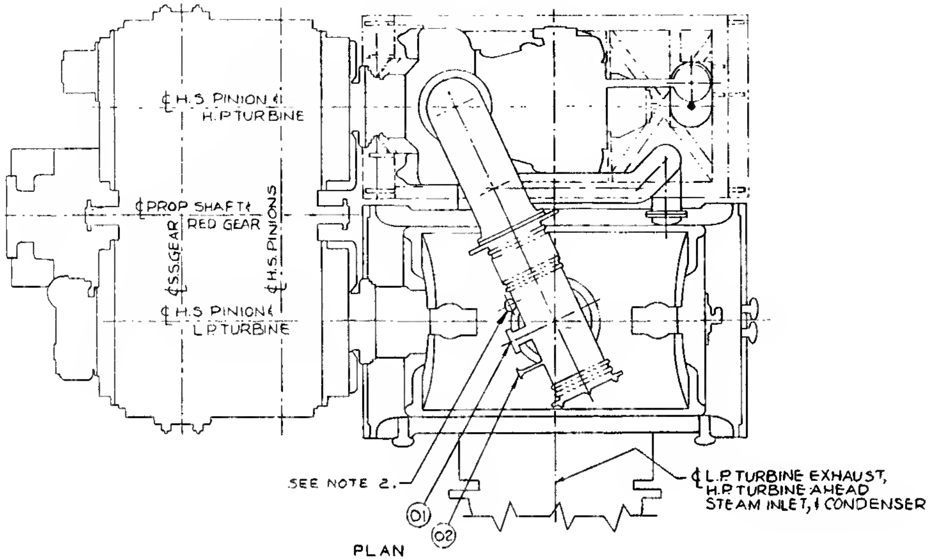
Bearing cover shock loads found in May were 419,000 lbs at the coupling end and 413,000 lbs at the thrust end (in the vertical direction). Jud's stress calculations, different from Dale Evans', showed that the allowable loads were 37,600 lbs at the coupling end and 199,000 lbs at the thrust end.

Jud next went back to the shock model to see if he might be able to decrease the loads by changing the structure of the bearing supports. He found that bearing loads increased both when the bearing supports were made softer and when they were made stiffer. This effect occurred because, at higher or lower rotor frequencies, the rotor would act as a "tuned" vibration absorber for other components of the model.

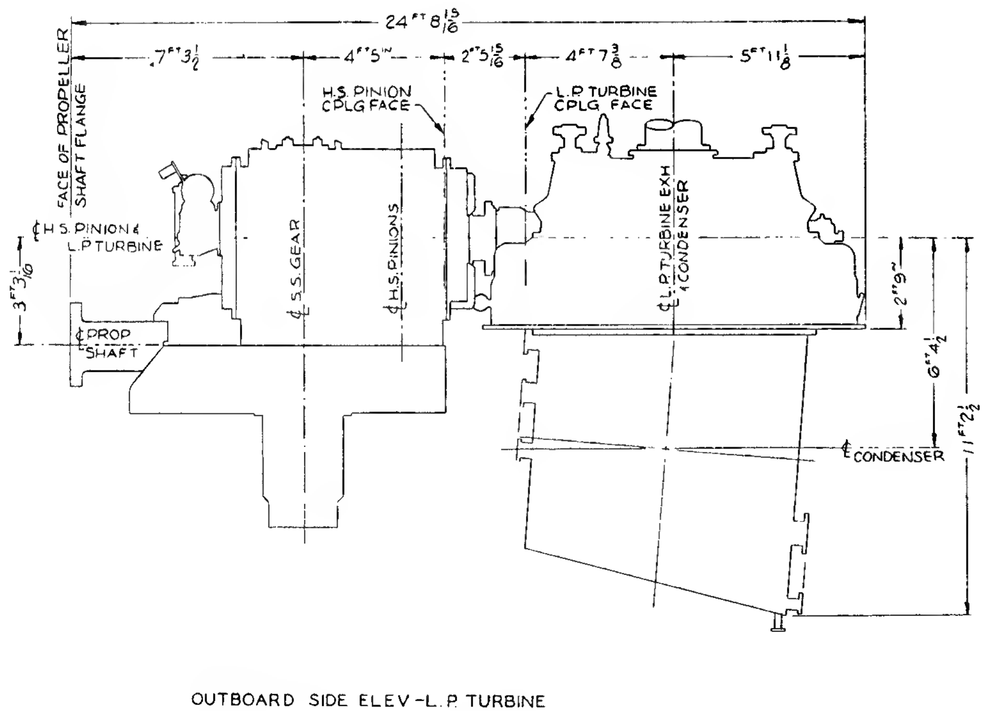
He also considered making the spherical keys much larger so that the load applied to the covers would be distributed rather than concentrated. After discussion with the other engineers working on the project this alternative was rejected for several reasons. First, the bearings had already been ordered. Second, the bearings had performed satisfactorily

in the past and it was felt that changing the design might in some way impair performance. They also expected that DOVE would be hostile to changes in a proven design. It was further thought desirable to keep the same bearing so spare parts would be the same for the new vessels as for the older destroyer escorts which had been built using the same LP turbine.

At this time the Production Department told engineering that they could not wait for new, stronger bearing covers to be designed and cast; the delay would throw construction of the first turbine too far behind schedule. Jud felt that it might be possible to make the existing bearing covers strong enough by welding ribs around their circumference.



H.P. EMERGENCY EXHAUST PIPE	408	
CROSSOVER PIPE	2,000	
6750 SQ FT CONDENSER (OPERATING)	42,000	
MAIN REDUCTION GEAR COMPL.	83,100	
L.P. TURBINE COMPL	33,385	
H.P. TURBINE COMPL (EXCLUDING GIRDER)	15,560	
TOTAL WT-MAIN PROP MACHINERY	175,160	
COMPONENT	ESTIMATE	ACTUAL
TABLE OF WEIGHTS		



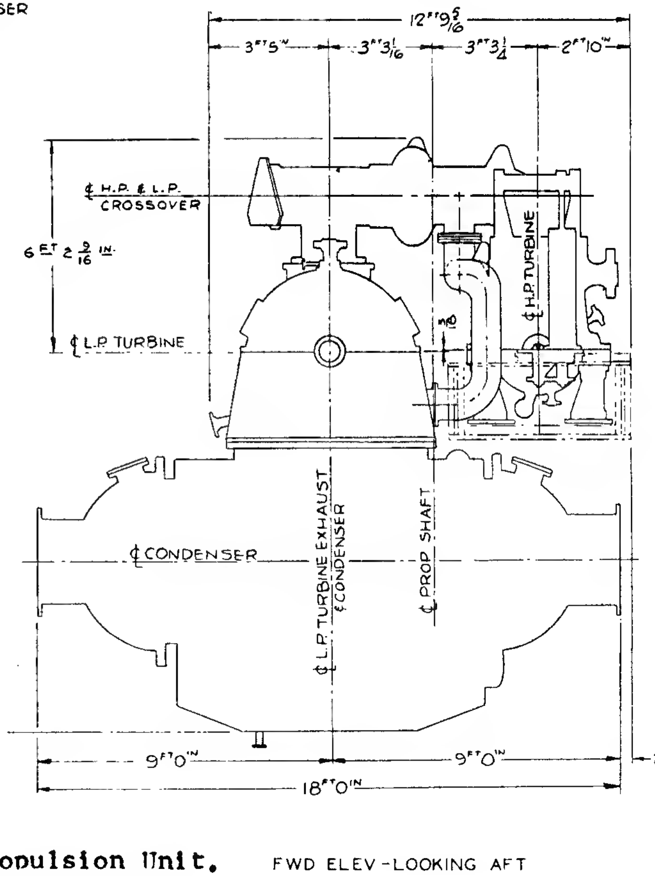


Exhibit 1: Outline Drawing of the Propulsion Unit.

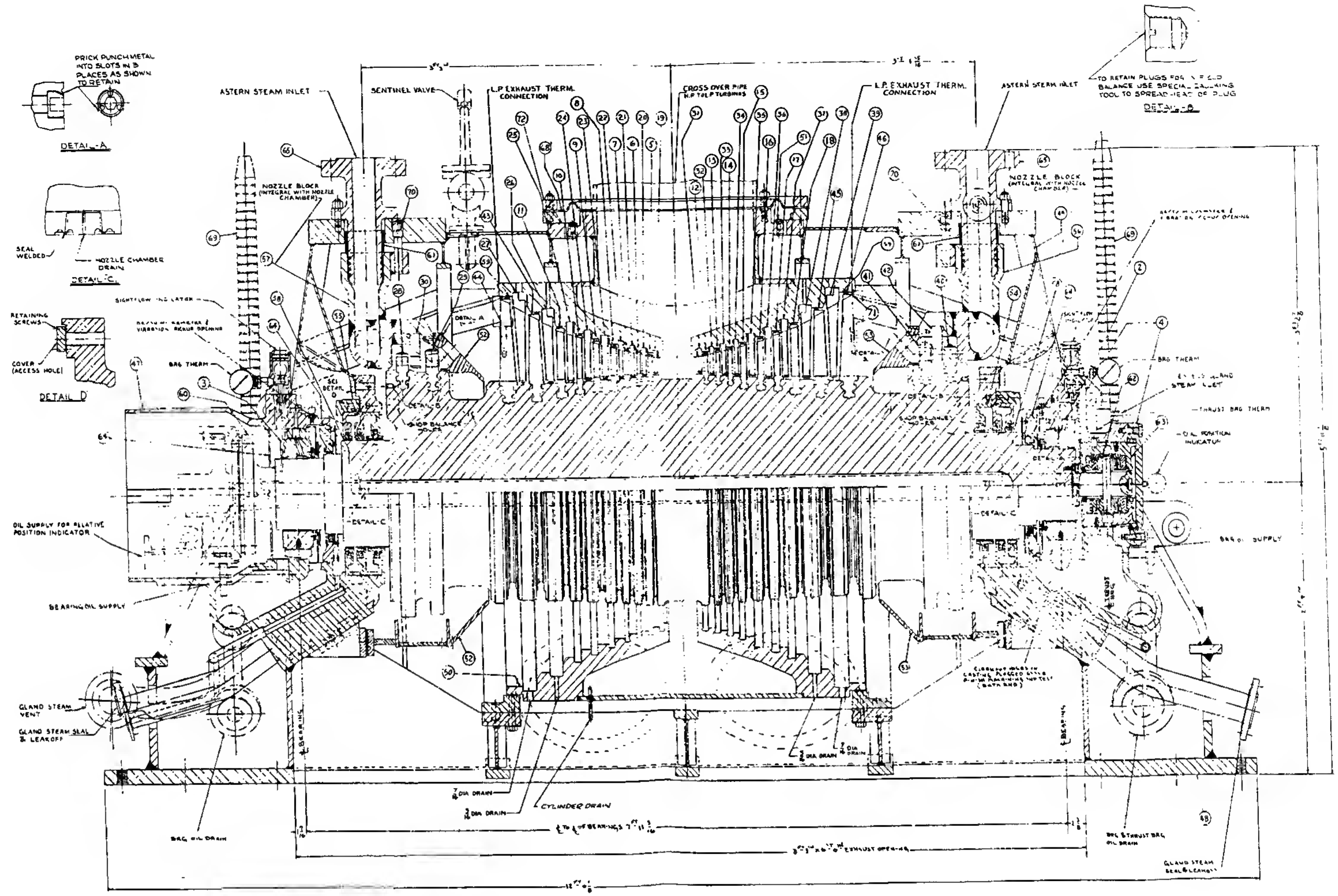


Exhibit 2: Assembly Drawing of the LP Turbine.

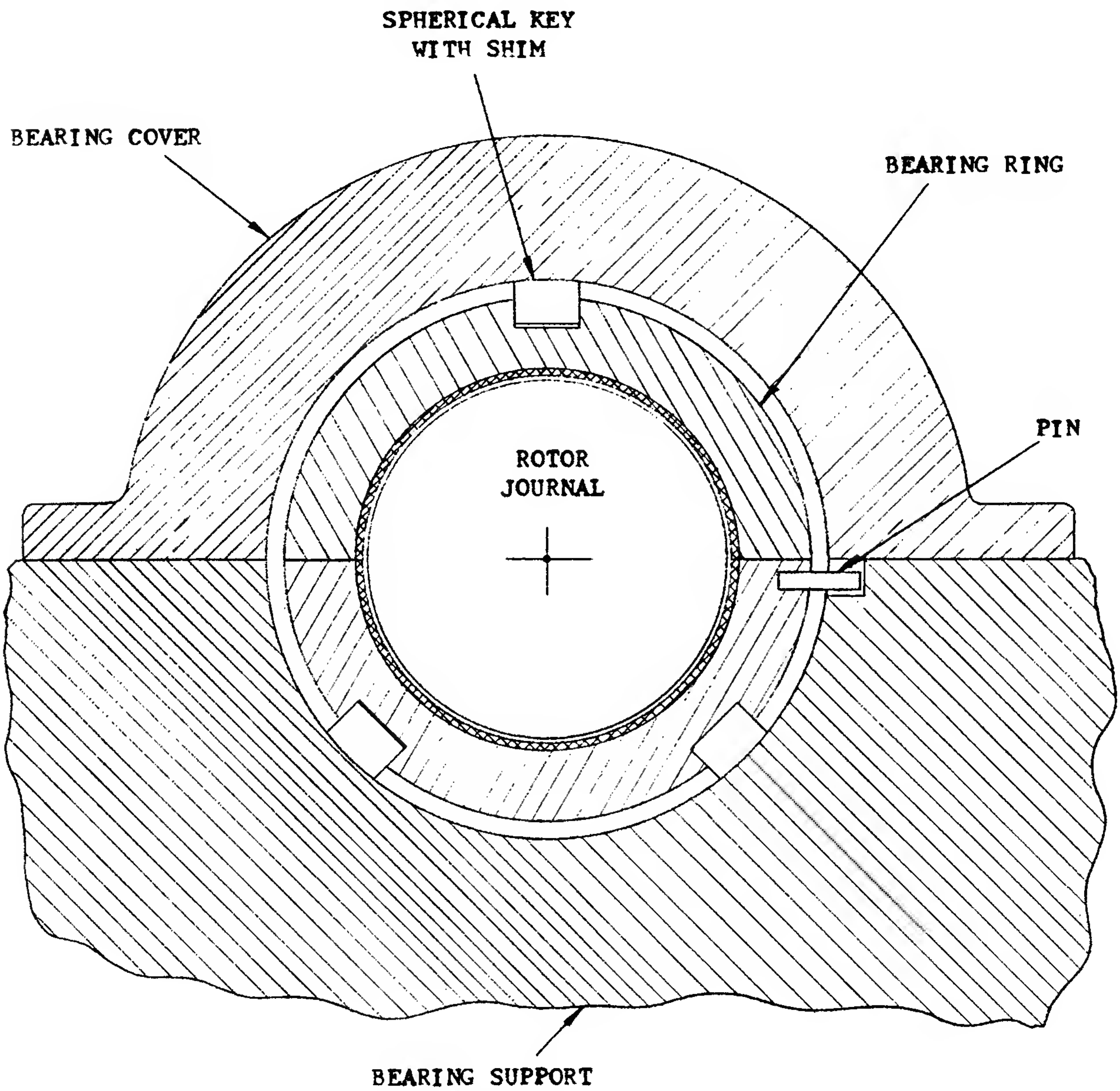


Exhibit 3: Section View Sketch Showing Spherical Keys. Not to Scale. Bearing Metal Relief Not Shown.

ECL 49-A
Exhibit 4

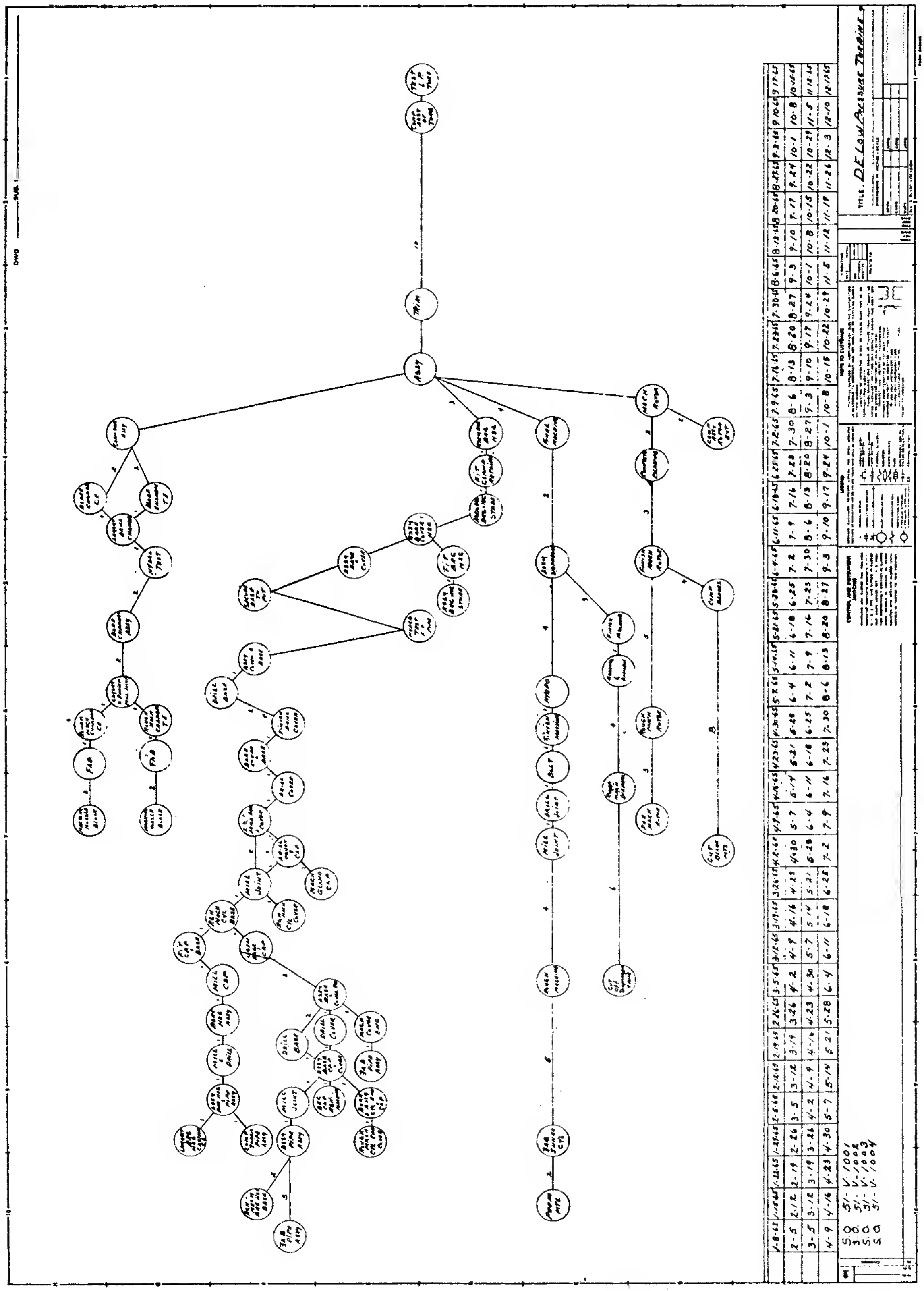


Exhibit 4: PERT Chart for Fabrication of the LP Turbine.

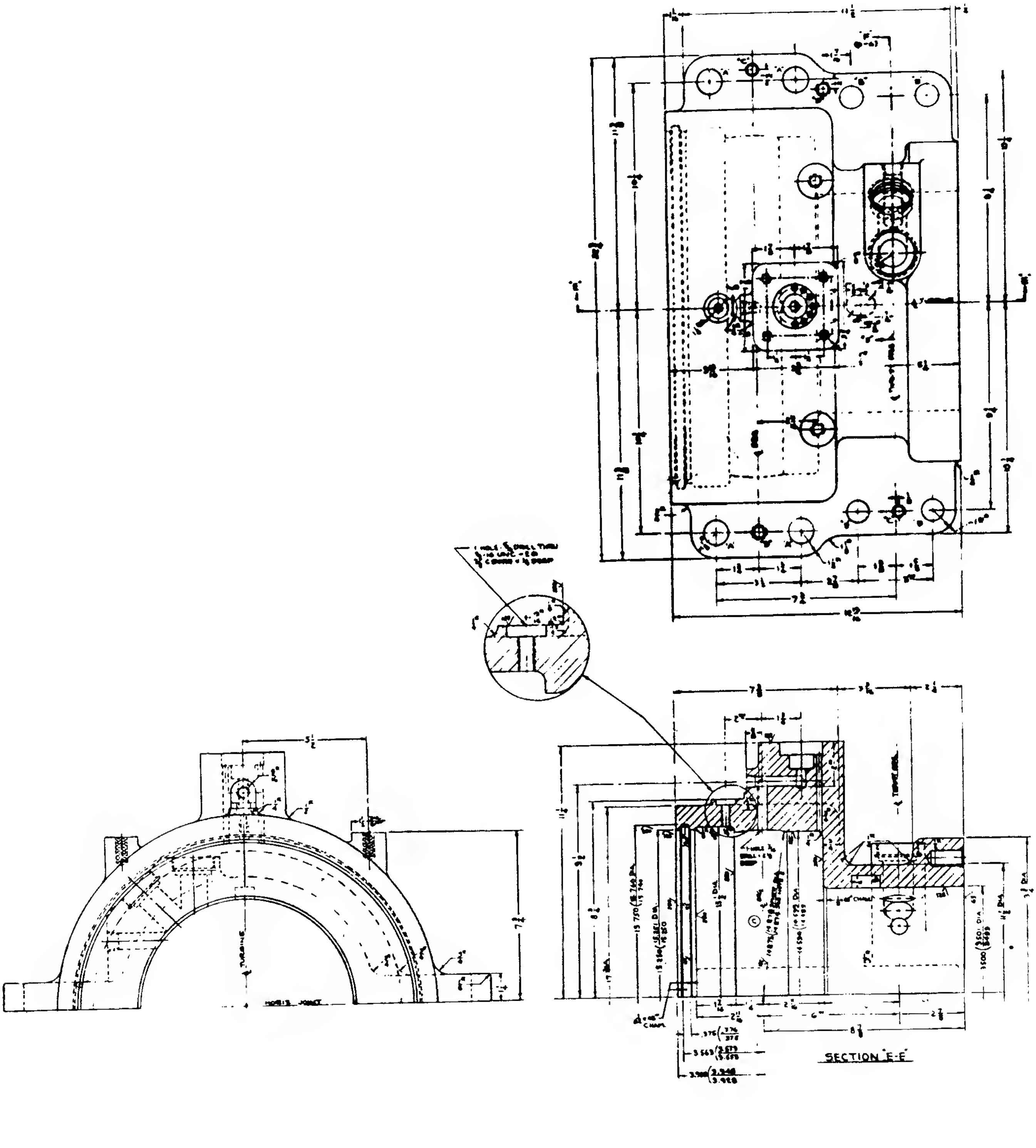


Exhibit 5: LP Turbine Thrust End Bearing Cover -- Original Design.

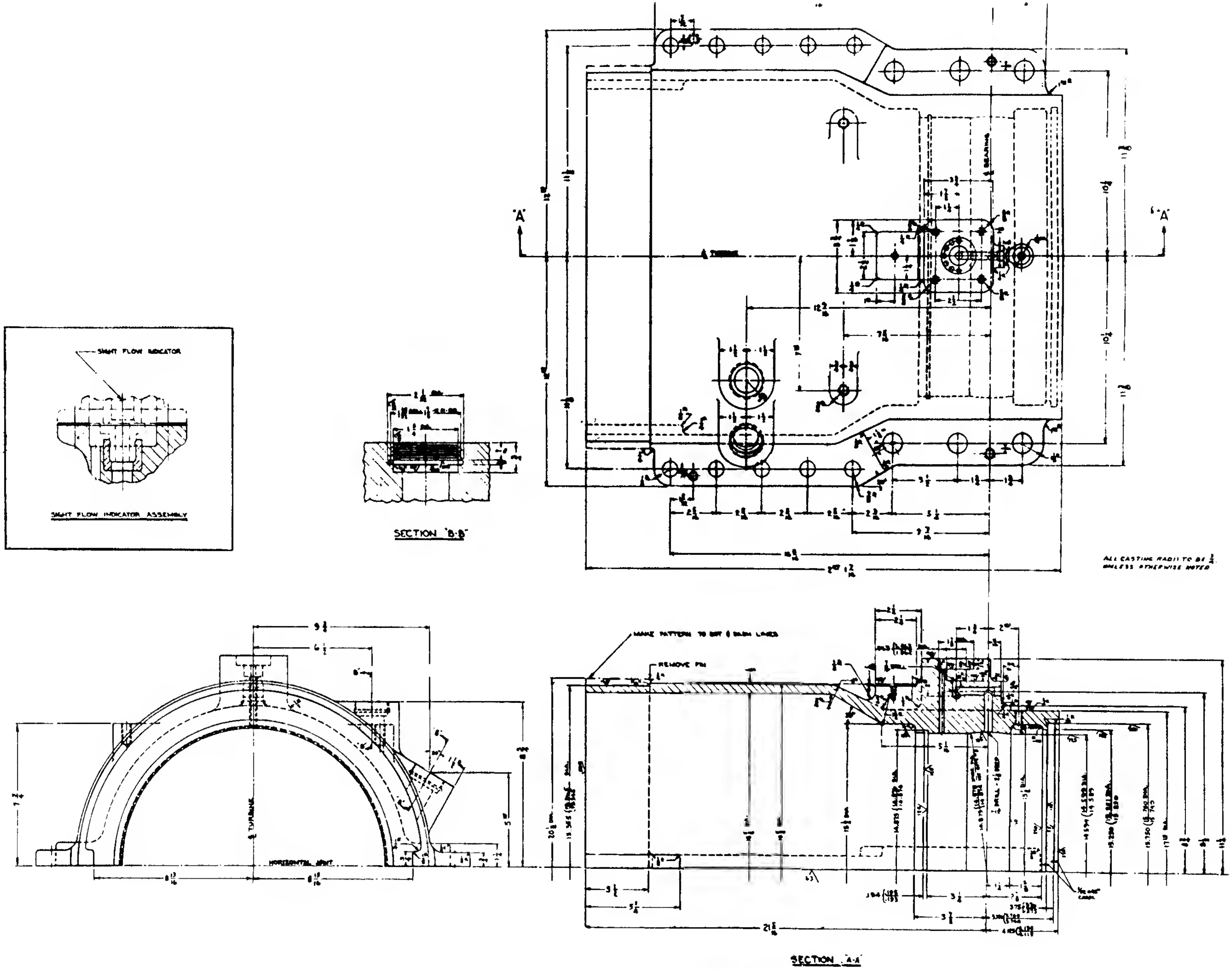


Exhibit 6: LP Turbine Coupling End Bearing Cover -- Original Design.

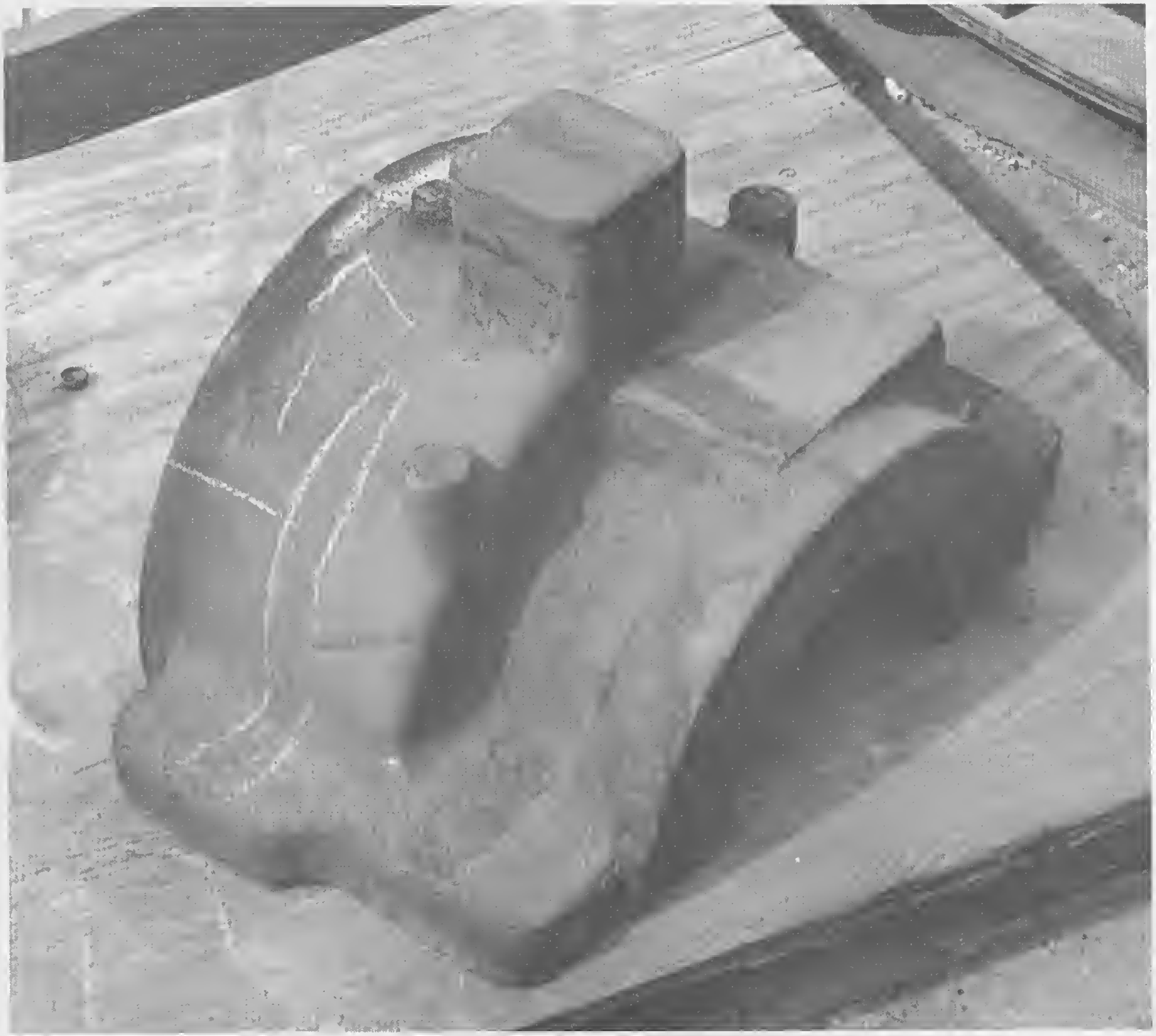


Exhibit 7: Original Design Thrust End Bearing Cover Casting Before Machining.

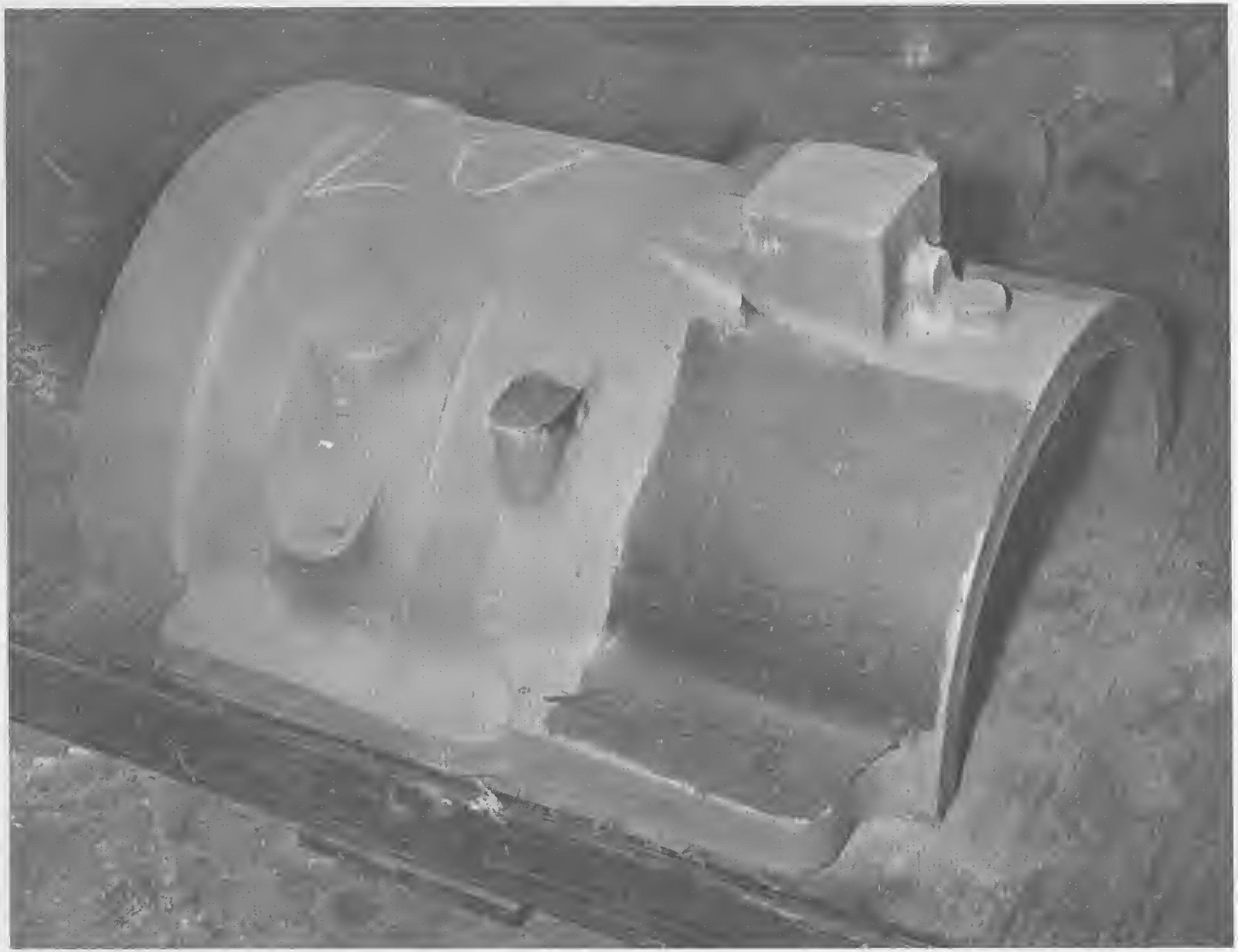


Exhibit 8: Original Design Coupling End Bearing Cover Castings Before Machining.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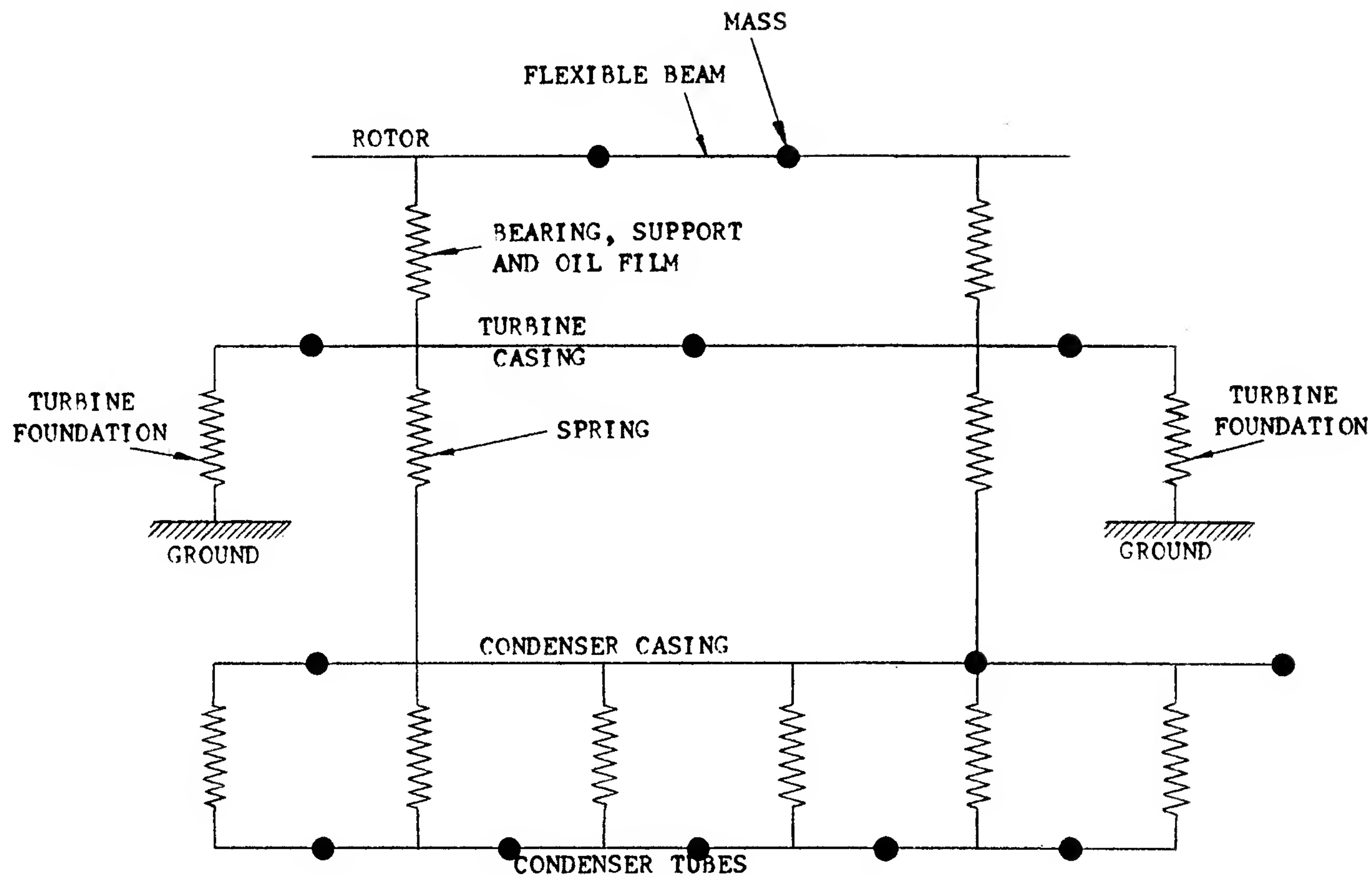


Exhibit 9: Sketch of LP Turbine Shock Model.

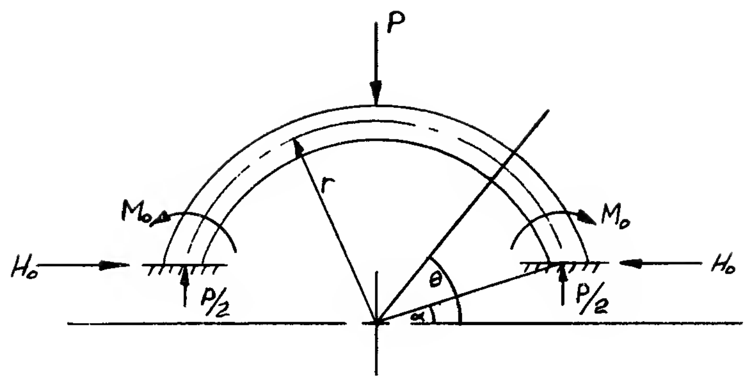
ZAMCON CORPORATION (B)

Design of a Steam Turbine Bearing Cover

Jud Noble thought that perhaps the LP turbine bearing covers could be strengthened enough to meet DOVE's shock design requirements by welding circumferential ribs to the castings. The Production Department had told him that waiting for new and stronger covers to be designed and cast would throw work on the first two turbines far behind schedule. Jud wanted to find out how much stronger the covers would be with added ribs and with gussets to strengthen the flanges in bending. The studs holding the covers to the bearing supports were also overstressed. Because the bearing supports had already been fabricated into the cylinder base assembly, the number of studs could not be increased. However, it would be possible to use larger studs of a higher strength material and to move them slightly nearer the shaft centerline by enlarging and shifting the centers of the existing holes.

Jud sketched longitudinal cross-sections of bearing covers with ribs placed in several locations and calculated their strengths. This was purely an investigatory step -- to see how much stronger the covers could be made by adding reinforcements. A summary of the allowable loads he had obtained by May 19th appears in Exhibit 1. All ribs were 7/8 inch wide by 2-7/8 inches high. While he found that a thrust end cover with two ribs would be acceptably strong, even with three ribs the strength of the coupling end cover was only a little more than half the design shock load (which Jud had rounded to 425,000 lbs for both covers).

Jud's calculations for the thrust end cover appear in Exhibit 2. The formula on page 1 of the exhibit for the bending moment M at any point in a section was obtained from a general formula for the moment in a "thin" partial ring (an arch) with built-in ends and a concentrated load P acting at its center. It appeared in an article by Alexander Blake entitled "How to Find Deflection and Moment of Rings and Arcuate Beams," in the January 7, 1963 issue of Product Engineering for the arch drawn below:



The bending moment at any angle θ between α and $\Pi/2$ is given as:

$$M = H_o r (\sin \theta - \sin \alpha) - \frac{Pr}{2}(\cos \alpha - \cos \theta) + M_o$$

Where:

$$H_o = P \frac{[8(1 - \sin \alpha) \cos \alpha - (\Pi - 2\alpha)(1 + \cos 2\alpha)]}{[2(\Pi - 2\alpha)(\Pi - 2\alpha + \sin 2\alpha) - 16 \cos^2 \alpha]}$$

$$M_o = H_o r \frac{[2(\Pi - 2\alpha + 2 \cos \alpha)(\sin \alpha - 1) + (\Pi - 2\alpha)^2 \cos \alpha]}{[8(1 - \sin \alpha) \cos \alpha - (\Pi - 2\alpha)(1 + \cos 2\alpha)]}$$

These equations are not explicitly derived in the article; it is merely stated that: "The equations were derived by use of Castigliano's theorem and expressions for strain energy due to bending." The formula for M which appears on Jud's calculation sheet was obtained from that above by substituting $\alpha = 0°$. He also used a correction factor with the equation to account for the bearing covers being "thick" rings rather than "thin." For the strength computations, the longitudinal cross-section of the thrust end cover was divided into four parts, plus the ribs, as shown on page 2 of Exhibit 2. Since at all sections the ratio of the radius of curvature to the thickness is less than 10, appropriate thick ring correction factors (depending on the ratio at the particular numbered section) from an article entitled "Stress in Rings" in the November 13, 1963 issue of Design News were applied.

For each numbered section: b = length, h = height, r' = mean radius, A = area, and I_o = moment of inertia about the centroid of that section. Using the formulas at the bottom of the page, Jud then computed the moment of inertia I of the entire acting cross section and its mean radius $\bar{r}$.

Allowable loads at the two angular positions in the cover are calculated on page 3 of Exhibit 2. Jud assumed that the flange on the cover was perfectly rigid, hence that the cap would act as if it were built in at $\theta = 10°$. He explained that he then knew that the highest stress would be either at $\theta = 90°$ (Position 1-1) -- the top of the arch, at the point of load application, or at $\theta = 10°$ (Position 2-2) -- where the cover was "built in" to the flange. The formulas at the top of page 3 already include the numerical value of the yield stress; c = the distance from the centroid of a section to the extreme fiber in the cover and k = the correction factor for a thick ring.

Bearing cover castings of the original design were already in process. Although they knew the coupling end bearing cover could not possibly meet the shock design criteria, management of the Marine Engineering Department felt that they had no choice but to go ahead and weld ribs onto covers of the old design for the first two turbines. Irving Ashby would try to get approval from DOVE for a design deviating from the specifications.

At this point Jud did not know if the 3-rib coupling end cover and the 2-rib thrust end cover would fit inside the turbine without interferences. He had a draftsman make layouts to determine the sizes and numbers of ribs that would fit. They found that there was room for only a single rib -- at the centerline of the bearing on each cover. Fortunately each rib could be 1-1/4 inch wide by 2-7/8 inches high -- 3/8 inch wider than the ribs Jud had first considered. The most critical clearance requirement was that room remain to remove the gland seal retainers and the shaft seals. D O V E requires that these seals be replaceable without dismantling the turbine. It was too late to redesign any of the other parts for more clearance.

Allowable loads before yielding for covers modified with a single rib were found to be 148,500 lbs at the coupling end and 148,800 lbs at the thrust end (for bending at the center of the cover). Jud's calculation sheet for the thrust end cover appears in Exhibit 3. This computation was made on June 23rd.

Note that he had now decided to neglect all portions of the cover having an inside radius of less than 7.30 inches, assuming that their contribution to the bending strength of the cover under a concentrated load application was negligible. Comparing this with his earlier calculation on page 2 of Exhibit 2, all of section 4 and most of section 3 would now be neglected. To obtain a consistent set of figures, Jud recomputed the strength of a ribless thrust end bearing cover neglecting these portions of the cover. He now found the allowable load to be 41,800 lbs rather than 199,000 lbs.

The shop was instructed to modify two sets of bearing covers by welding a single rib plus several flange gussets to each and enlarging and relocating the holes for stronger studs. Irving did not immediately ask D O V E to approve the modified cover design for the first two

turbines; he wanted to wait and see if there would be strength problems in other areas. Jud's shock analysis team had not yet completed the mathematical models for finding athwartships and fore and aft shock loads.

As soon as the decision to modify two sets of the original covers was made, Jud began working with draftsmen to design new cast covers to be used on the 24 remaining turbines. He wanted to determine where metal could be placed to make the covers as strong as possible and still preserve the necessary clearances. Jud sketched alternative cover configurations and calculated stresses, then he picked the strongest that would fit the available space. At the same time metallurgists in the materials department were investigating stronger casting materials. A steel with a yield strength of 50,000 psi was chosen to replace the old material, which had a yield strength of only 30,000 psi. By early July draftsmen had completed new bearing cover drawings from Jud's sketches. At first they worked on layouts. Jud checked these to make sure they were in accord with his sketches and would be strong enough. The Purchasing Department sent the layouts to the vendor foundry. The draftsmen later made detailed drawings of the new covers. These are shown in Exhibits 4 and 5. Photographs of the new covers appear in Exhibits 6 and 7. A summary of the stresses in all three sets of covers appears in Exhibit 8. Jud's strength calculations for the new thrust end cover are shown in Exhibit 9.

Working from the layouts they had received, the vendor foundry modified the patterns for the original covers to conform to the new design and delivered new cast covers to Zamcon by the end of July. This was much earlier than had been expected. A week before the new castings arrived, Irving discovered that they would soon be ready. When he checked with the Production Department, he found that the shop had not yet begun to modify the original covers. At this point it was obviously best to wait for the new castings, so the work orders for the modifications to the old covers were cancelled. Irving thus never had to ask DOVE to approve the reinforced original design covers.

Work on the mathematical model for the LP turbine did not cease in May. Expansion of the capabilities of the computer program and concurrent refinement of the model continued. Because the computer program is used only to find the normal modes, the other steps in the shock analysis being performed by the engineers, and because of the waiting periods for computer time, several weeks usually elapse between a change in the model -- different masses or spring rates -- and final results in the form of loads or stresses.

In September 1965 Jud Noble proposed to DOVE a change in the modeling of the condenser, which is hung from the LP turbine. The condenser tubes, which had been considered simply supported, were now to be treated as having built-in ends, a more realistic representation. Prior to this time the computer program could not accomodate moments acting on the beams. The model, as Jud drew it to show DOVE, appears in Exhibit 10, complete with the numerical values used. The layout

of this model is identical with that of Exhibit 9 of Part A. The flexural rigidity of each beam is written below the beam and the distance between each of the twelve numbered segments above that segment of the beam (note that the model drawings are not to scale). Spring constants appear next to each spring. Below each mass appears its mass value. The letter E represents the base 10, the following number being the exponent, thus E6 = 10^6. The moment restraints of 2.5×10^{11} in-lb/rad at each end of the condenser tube bundle are noted just below the spring constants for the supports at the ends of the tubes. DOVE approved treating the condenser tubes as built-in and this change "detuned" the rotor and tubes, resulting in a reduction of the calculated bearing cover shock loads from 425,000 lbs to about 275,000 lbs.

CALCULATION SHEET
FORM 33752

REFER TO	FIGURES BY	DATE
		5/19/65

Summary of Brg. Cap. Stress's - L.P. TURBINE

	CPLG END	THRUST END
YEILD LOAD - CENTER OF BRG CAP		
As DESIGNED	T 37.6 $\times 10^3$ lbs	199T/191C $\times 10^3$ lbs
w/ 1 rib	T 98.2 "	353T/259C "
w/ 2 rib	T 173. "	485T/298C "
w/ 3 rib	T 232. "	
YEILD LOAD - Junction of FLANGE & SHELL		
As DESIGNED	242 $\times 10^3$ lbs	1283T/1229C $\times 10^3$ lbs
W/ 1 RIB	618 "	2276T/1666C "
W/ 2 RIB	1030 "	3124T/1918C "
W/ 3 RIB	1493 "	
YEILD LOAD - FLANGE BENDING AT STUD HOLES As DESIGNED	26.8 $\times 10^3$ lbs	26.8 $\times 10^3$ lbs
W/ Ribs	—	—
YEILD LOAD - FLANGE STUDS 1" 8 UNC - Mil B857 Gr5 As DESIGNED	39.6 $\times 10^3$ lbs	39.6 $\times 10^3$ lbs
W/ RIBS (TENS. ONLY)	233. " (3 ribs)	186. $\times 10^3$ lbs (2 rib, 1 gusset)
Stud Pull-out From Mating Flange % of Stud Lgn Req'd, + Status	74% - OK	74% - OK
Req'd Design Load - approx	425. $\times 10^3$ lbs	425 $\times 10^3$ lbs
Mil B857 STUD Size Req'd for Design Load Tensile Load Only - No flange Magnification	1¼ Gr 8 1½ Gr 5	1¼ Gr 8 1½ Gr 5

Exhibit 1: Allowable Bearing Cover Loads with Added 7/8 Wide x 2 7/8 High Ribs.

SUBJECT	NO. SHEETS	SHEET NO.
LP Turb. Brg Cap Stress		

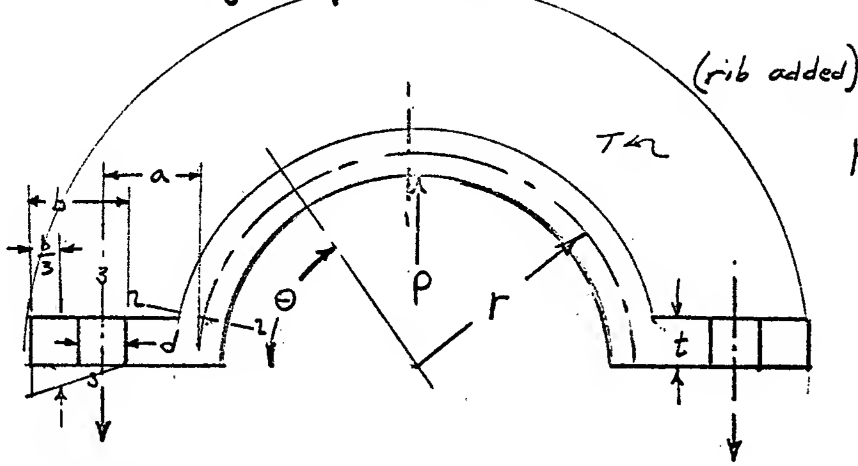
CALCULATION SHEET
FORM 33752

REFER TO	FIGURES BY	DATE
		5/18/65

Exhibit 2: Thrust End Bearing Cover Strength Calculations.

Revised - 6/23/65

L.P. Turb Brg Cap - Allowable Load



Mat'l:
Y.S. = ~~30K~~ 50 Kpsi
U.S. = ~~60K~~ 80 Kpsi
E = 30×10^6

Cap Bending
Ref: Prod Engrg, 7 Jan 63

Assume Built-In Arch, 180° included angle

$$M = -\frac{Pr}{2}\left(.918 \sin\theta + \cos\theta - 1.221\right)$$

Stress

$$\sigma = \frac{M}{I/c} \qquad M = \sigma \frac{I}{c}$$

ref:

$$\sigma \frac{I}{c} = -\frac{Pr}{2}\left(.918 \sin\theta + \cos\theta - 1.221\right)(\text{correct. factor for thk. ring}) = f\left(\frac{r}{c}\right)$$

Flange Bending - assume 0 bending due to rib support

~~$M = \frac{P}{2}a$ $\sigma = \frac{M}{I/c}$ I - allow for bolt holes~~

Bolt Load $$T = \frac{P}{2}\frac{\left(a + \frac{2}{3}b - d/2\right)}{\left(\frac{2}{3}b - d/2\right)}$$

$$\sigma = \frac{T}{A_{bolt(root)}}$$

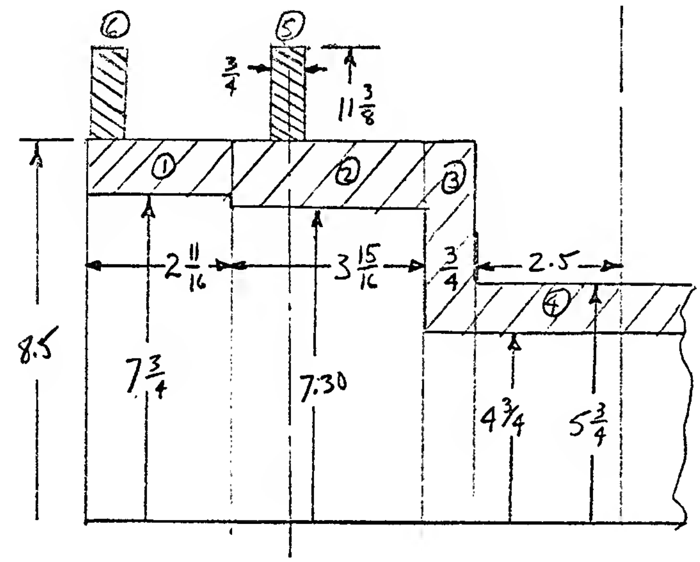
SUBJECT		NO. SHEETS	SHEET NO.
Bearing Cap Stress	LP TURBINE		

CALCULATION SHEET
FORM 33752

REFER TO	FIGURES BY	DATE
		5/19/68

THRUST END

CAP BENDING



REF 715J149
Mat'l
(stl. Cast'g)
Y.S. = 30 Kpsi
T.S = 60 Kpsi

Max Eff = 5x thk. shell at Brg = 5x 1.2 = 6" either side of brg ℄

Section Properties of Arch

Item	b	h	r'	A	Ar'	Ar'²	Io	①–④	①–③
1	2.69	.75	8.12	2.02	16.39	133	.14		
2	3.94	1.20	7.90	4.73	37.35	295	.85		
3	.75	3.75	6.63	2.81	18.64	124	4.94		
4	2.5	1.0	5.25	2.50	13.12	69	.20	$\bar{r}$ = 7.09	
Σ				12.06	85.50	621	6.1	I = 20.8	
Σ ①②③				[illegible]	[illegible]	2			
5	.75	2.88	9.94	2.16	21.47	213.	2.24	$\bar{r}$ = 7.52	
Σ				14.22	107.0	834	8.4	I = 37.3	
6	.75	2.88	9.94	2.16	21.47	213.	2.24	$\bar{r}$ = 7.84	
Σ				16.34	128.4	1047	10.6	I = 51.0	

$$CG = \bar{r} = \frac{\Sigma Ar'}{\Sigma A}$$

$$I = \Sigma I_o + \Sigma Ar'^2 - (\Sigma A)\bar{r}^2$$

SUBJECT	NO. SHEETS	SHEET NO.

CALCULATION SHEET
FORM 33752

REFER TO	FIGURES BY	DATE
		5/19/65

Thrust End (Cont)

Load at Yeild

$$\theta_{1-1} = 90°, \quad P = 198 \frac{I}{\bar{r}ck} \times 10^3 \text{ lbs} \quad (Y.S. = 30k)$$

$$\theta_{2-2} = 10°, \quad P = 1275 \frac{I}{\bar{r}ck} \times 10^3$$

Condition	$\bar{r}$	c	$\bar{r}/c$	k	$\bar{r}ck$	$I/\bar{r}ck$	P_{1-1} YEILD	P_{2-2} YEILD	
As Designed	7.09	4.29	1.65	.69	20.68	1.006	199.2×10^3	1283×10^3	Tension
	7.09	2.34	3.03	1.30	21.57	.964	190.9×10^3	1229×10^3	Compression
w/rib 5	7.52	3.86	1.95	.72	20.90	1.785	353.4×10^3	2276×10^3	T
	7.52	2.77	2.71	1.37	28.54	1.307	258.8×10^3	1666×10^3	C
w/rib 5+6	7.84	3.54	2.21	.75	20.82	2.450	485.1×10^3	3124×10^3	T
	7.84	3.09	2.54	1.40	33.92	1.504	297.8×10^3	1918×10^3	C

Flange Bending at Bolt Line – Without Ribs

$$\sigma \frac{I}{c} = \frac{1}{2} \times \frac{P}{2} \times a \; : \; P = \frac{4\sigma}{a}\left(\frac{I}{c}\right) = \frac{4 \times 30 \times 10^6}{(10.25 - 8.12)}\left(\frac{1}{6}\right)\overset{1.82}{(2.88 - 1.06)}(1.25^2)$$

$$= \frac{20}{2.12} \times 1.82 \times 1.562 \times 10^3 = 26.8 \times 10^3$$

Bolting – Without Ribs – Same as Cplg End $P_{yeild} = 39.6 \times 10^3$

Bolting – With Rib – Same as Cplg End except each bolt take 50% of Load for side

$$P_{yeild} = 233 \times 10^3 \left(\frac{40}{50}\right) = 186\,000 \text{ lbs}$$

Stud Pullout Load – Same as cplg end.

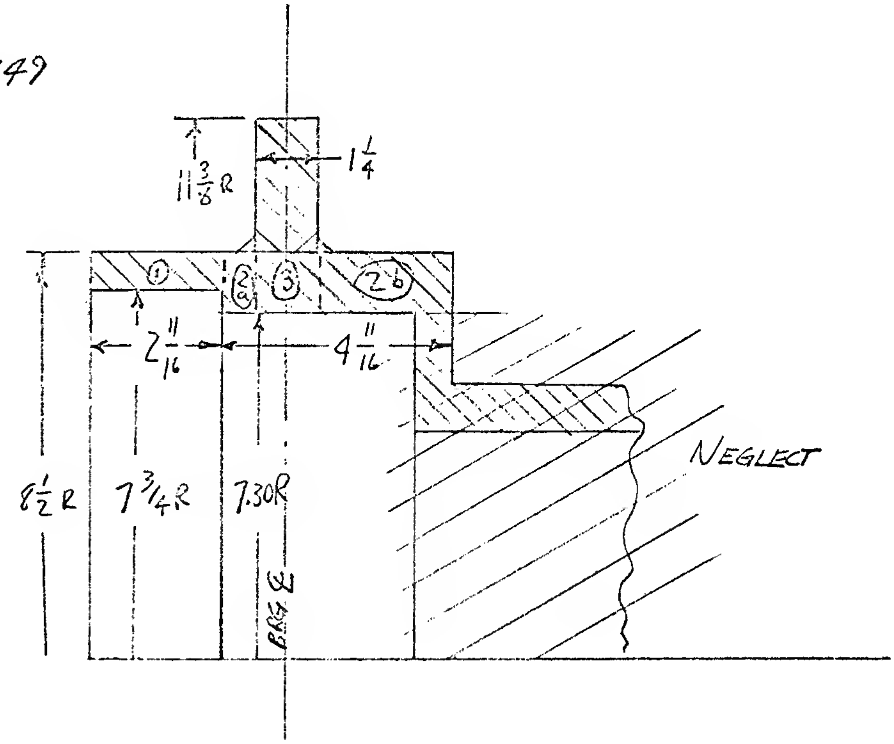
SUBJECT	NO. SHEETS	SHEET NO.

CALCULATION SHEET
FORM 33752

REFER TO	FIGURES BY	DATE
		6/23/65

Modified Casting 715J149
Thrust End
Brg Cap Bending

Mat'l:
Costing: Mil S15083 Cl B
Y.S = 30,000
Rib: PDS 5010
Y.S = 32,500



Item	b	h	r'	A	Ar'	Ar^2	I_o	$\bar{r}$	I	c	k	$\sigma_{\theta=90}$	$\sigma_{\theta=10}$	P-Yeild-90°
1	2.69	.75	8.12	2.02	16.38	133.02	.09							
2	3.44	1.20	7.90	4.13	32.61	257.63	.50							
3	1.25	4.08	9.34	5.10	47.63	444.90	7.07							
Σ				11.25	96.62	835.55	7.66	8.59	13.48					
Tension – top of rib										2.79	.81	92,800	23,600	148,800
Compression – inner surface										1.29	1.11	58,800	14,900	217,000

$$\bar{r} = \frac{\Sigma Ar'}{\Sigma A} \qquad I = \Sigma I_o + \Sigma Ar^2 - \bar{r}^2(\Sigma A)$$

$$\sigma = -\frac{Prck}{2I} \times \begin{pmatrix} -.303 \ @\theta = 90^\circ \\ -.077 \ @\theta = 10^\circ \end{pmatrix} \qquad P = 425\,000 \text{ lbs}$$

$$P_{yeild} = 425\,000 \times \left(\frac{\sigma_{YS}}{\sigma_{calc}}\right)$$

Exhibit 3: **Strength Calculation for "Modified Orig. Design" Thrust End Bearing Cover with Single Rib.**

SUBJECT	NO. SHEETS	SHEET NO.
LP Turbine Brg Caps		

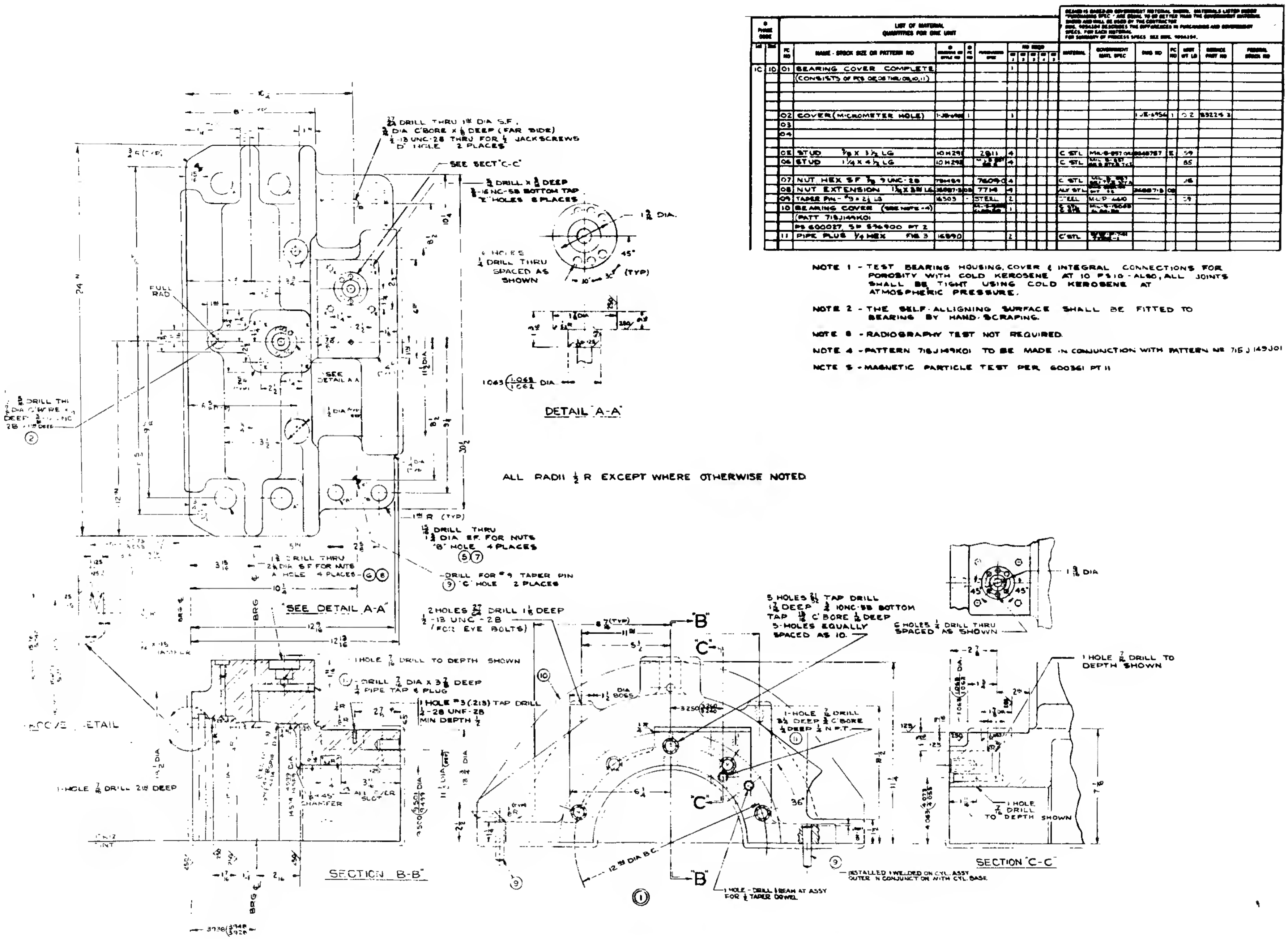


Exhibit 4: LP Turbine Thrust End Bearing Cover -- New Design.

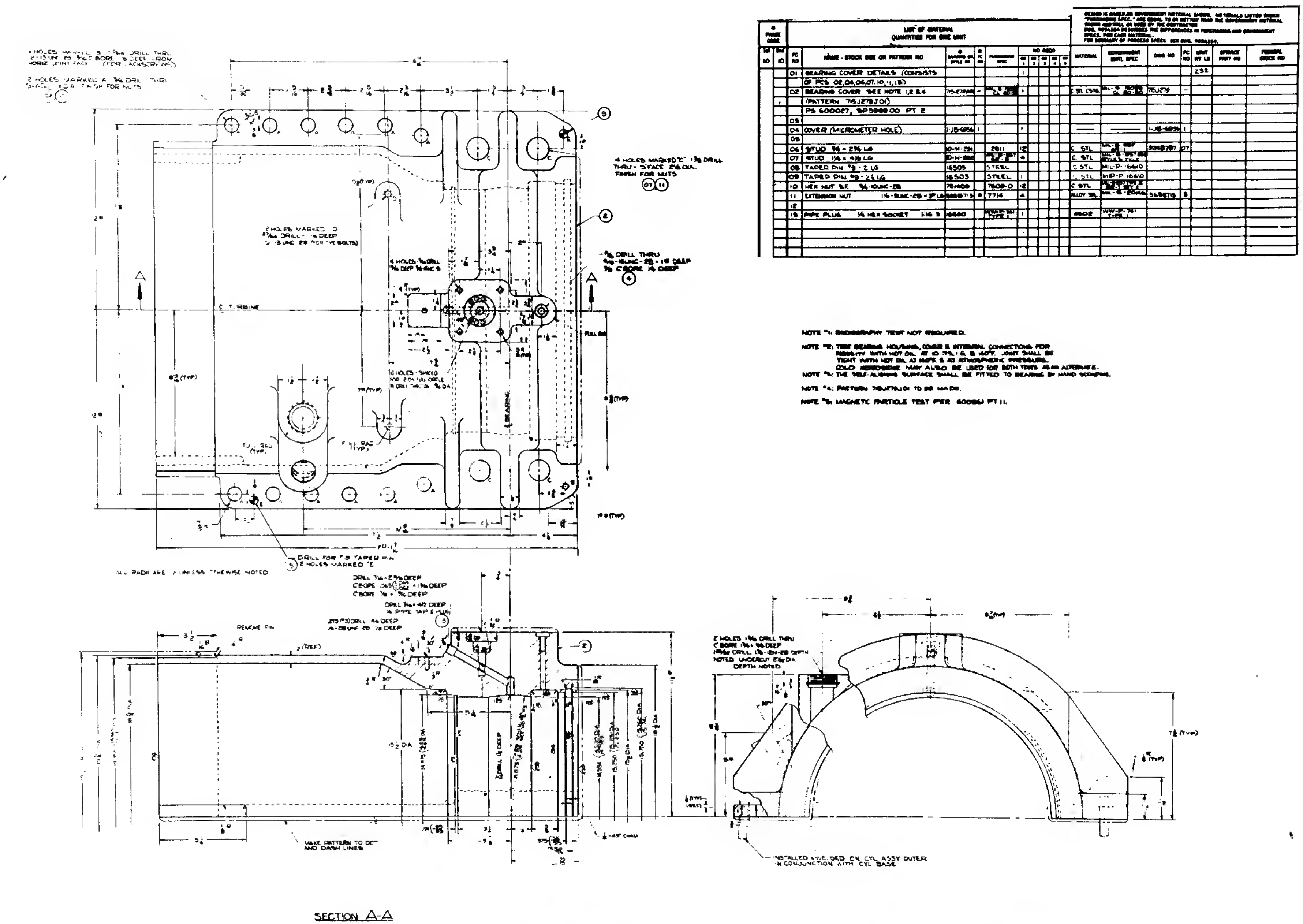


Exhibit 5: LP Turbine Coupling End Bearing Cover -- New Design.

Exhibit 6: New Design Thrust End Bearing Cover with Machining Completed.

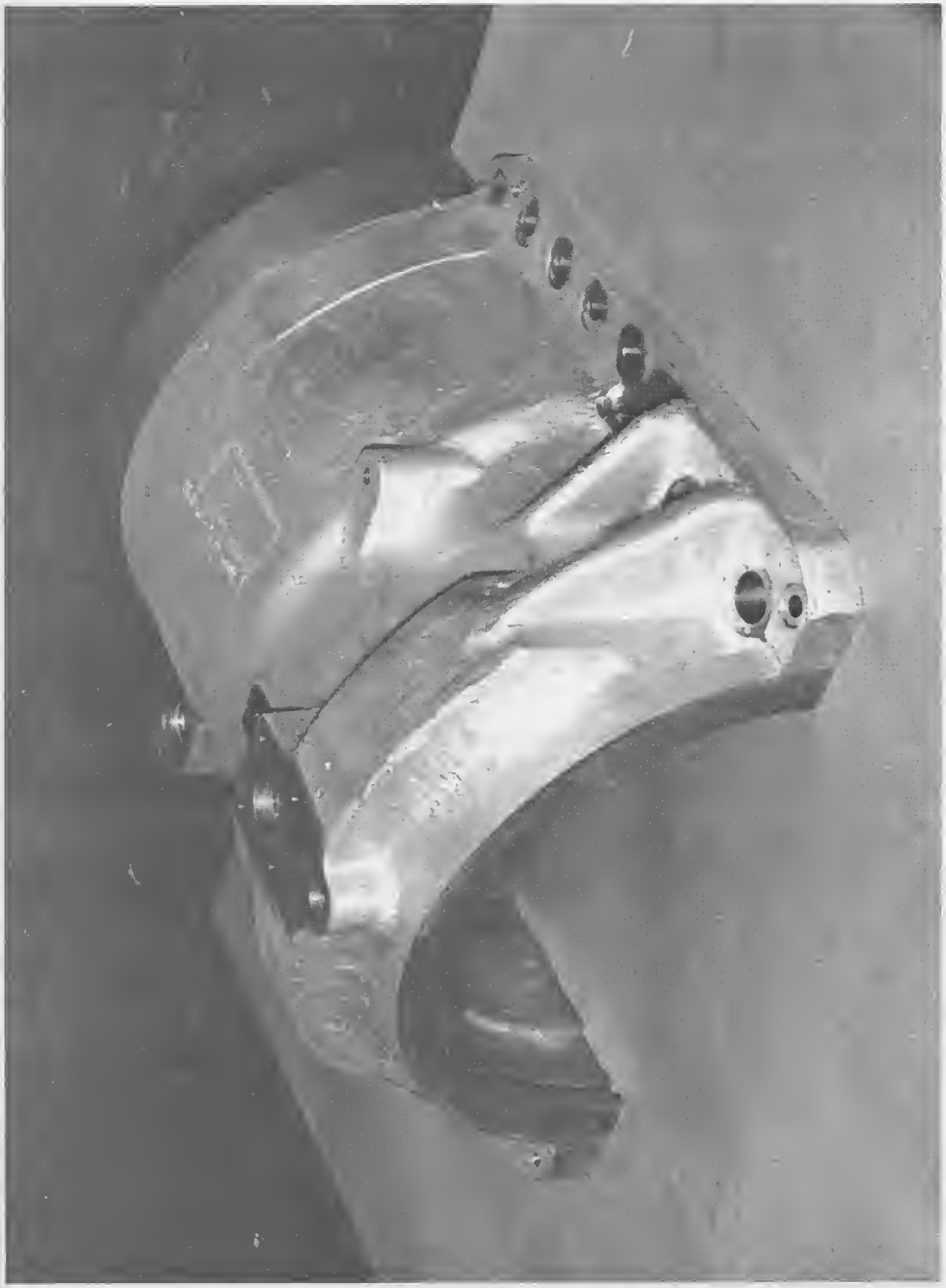


Exhibit 7: New Design Coupling End Bearing Cover with Machining Completed.

ENGINEERING CALCULATION REPORT

Exhibit 8 Summary of Bearing Cover Strengths. "Modified Orig. Design" Covers are Those with One Added Rib.

	Original Design.		Modified Orig. Design		New Design	
Materials	Cplg End	Thrust End	Cplg Eng	Thrust End	Cplg End	Thrust End
Cap Casting - Mat'l						
YS/TS $\times 10^3$ psi	30/60	30/60	30/60	30/60	50/80	50/80
Rib - Mat'l	—	—	PDS 5010	PDS 5010	Part of	Part of
YS/T.S $\times 10^3$ psi	—	—	32.5/65	32.5/65	Cstg	Casting
Stud Mat'l-Mil B857	Gr 5	Gr 5	Gr 8	Gr 8	Gr 8	Gr 8
Y.S./T.S $\times 10^3$ psi	78/115	78/115	120/150	120/150	120/150	120/150
Size	1"-8	1"-8	$1\frac{1}{4}$-8	$1\frac{1}{4}$-8	$1\frac{1}{4}$-8	$1\frac{1}{4}$-8
Brg Shock Design Load lbs $\times 10^3$	425	425	425	425	425	425
Equiv. 'G's'	71	71	71	71	71	71
Cap- BENDING AT Top						
Stress at design load - kpsi	339.	305.	93.7	92.8	45.7	48.5
Yeild Load - kips	37.6	41.8	148.5	148.8	465.	438.
'G's to yeild	6.3	7.0	25.	25.	77.7	73.2
Cap- bending at flange junction						
σ design load - kpsi	52.7	47.4	23.8	23.6	Low	Low
P yeild - KIPS	242.	269	—	—		
G's to yeild	40.	44.8	over 71	over 71	over 71	over 71
Flange-bending at stud ℄						
σ @ design load - kpsi	475.	475.	0	0	0	0
P yeild - kips	26.8	26.8	(No bending)			
G's to yeild	4.5	4.5				
Flange - shear under nut						
σ @ design load - kpsi	Not Checked	Not Checked	14.4	14.4	12.0	12.0

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ENGINEERING CALCULATION REPORT

	Original Design		Modified Orig. Design		New Design	
	Cplg End	Thrust End	Cplg End	Thrust End	Cplg End	Thrust End
Studs						
σ @ design load - kpsi	837.	837.	186.	181.8	114.5	114.5
$P_{proof\ load}$ - kips	39.6	39.6	274.	280.	445.	445.
G's to proof load	6.6	6.6	45.8	46.8	over 71	over 71
Stud - stripping in turbine base						
$\frac{\text{Design Load}}{\text{Stripping Load}}$ - %	133	133	85	85	85	85

FRAME	TITLE	DATE 6/24/65 REV. 7/7/65
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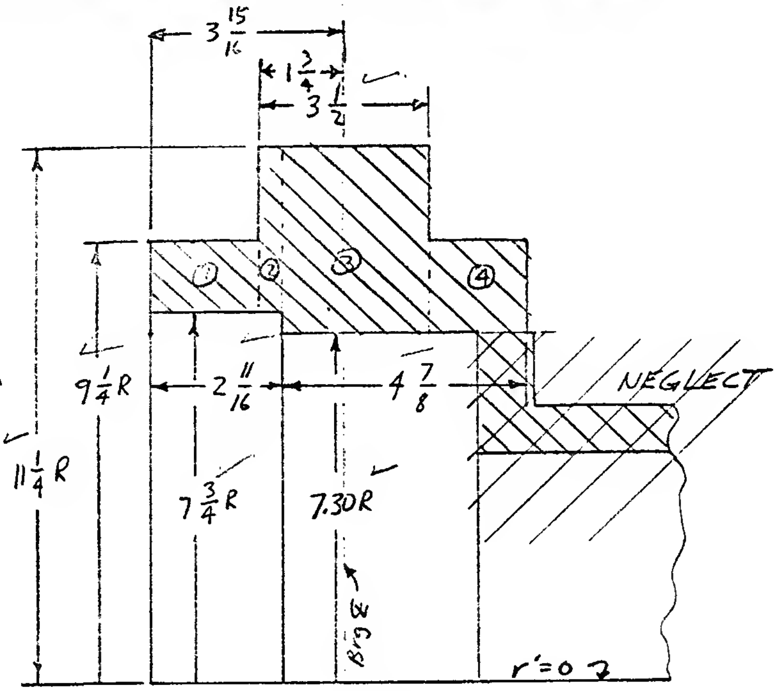
		PAGE 2 OF 2 PAGES

ENGINEERING CALCULATION REPORT

Exhibit 9: **Strength Calculations for New Design Thrust End Bearing Cover.**

New Casting - L 6077
Thrust End
Bearing Cap Bending

Mat'l: Mil S 15083 Cl. 80-50
Y.S. = 50 kpsi



Basic Section

Section Properties

Item	b	h	r'	A	Ar'	Ar'^{2}	I_o	$\bar{r}$	I	c	k	$\sigma_{\theta=90}$	Pyeild
1	2.188	1.50	8.50	3.282	27.897	237.12	.615						
2	.500	3.50	9.50	1.750	16.625	157.94	1.786						
3	3.00	3.95	9.275	11.850	109.909	1019.40	15.407						
4	1.875	1.95	8.275	3.656	30.255	250.36	1.158						
Σ				20.538	184.686	1664.82	18.966	8.992	23.02				
Tension - Top of R.L										2.26	.85	48,300	440 000
Compr - inner Surface										1.69	1.14	48,500	438 000

$$\bar{r} = \frac{\Sigma Ar'}{\Sigma A} \qquad I = \Sigma I_o + \Sigma Ar'^2 - \bar{r}^2(\Sigma A)$$

$$\sigma = -\frac{Prck}{2I} \times (-.303 \text{ @}90^\circ) \qquad P = 425000$$

$$P_{yeild} = P \times \frac{\sigma_{y.s}}{\sigma_{calc.}}$$

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ENGINEERING CALCULATION REPORT

New Castings

Flange Bending: Assume = 0 since bolt ℄ is inside a line between end of rib at brg ℄ and outside surface of shell at end of shell – i.e. load is taken by ribs in tension

Flange Shear under nut:

$$\tau = \frac{\frac{P\text{-lbs}}{2\text{ sides} \times 2\text{ studs}}}{\pi \times d \times t} = \frac{425\,000}{4 \times \pi \times 1.875 \times 1.5} = 12\,000 \text{ psi}$$

$$\tau_{allowable} = 50000/\sqrt{3} = 28{,}800 \text{ psi}$$

Studs – Assume 2-1¼ studs on each side take load equally

Stress Area: Mil B 857 Studs – Min area = thd root area = .929 in².

The stud is assumed to be on the centroid of the section – therefore there is no stress increase due to offset

$$\sigma = \frac{P}{A} = \frac{425000}{4 \times .929} = 114{,}500 \text{ psi}$$

$$P_{yeild} = 425\,000 \times \frac{120\,000}{114{,}500} = 445{,}000 \text{ lbs}$$

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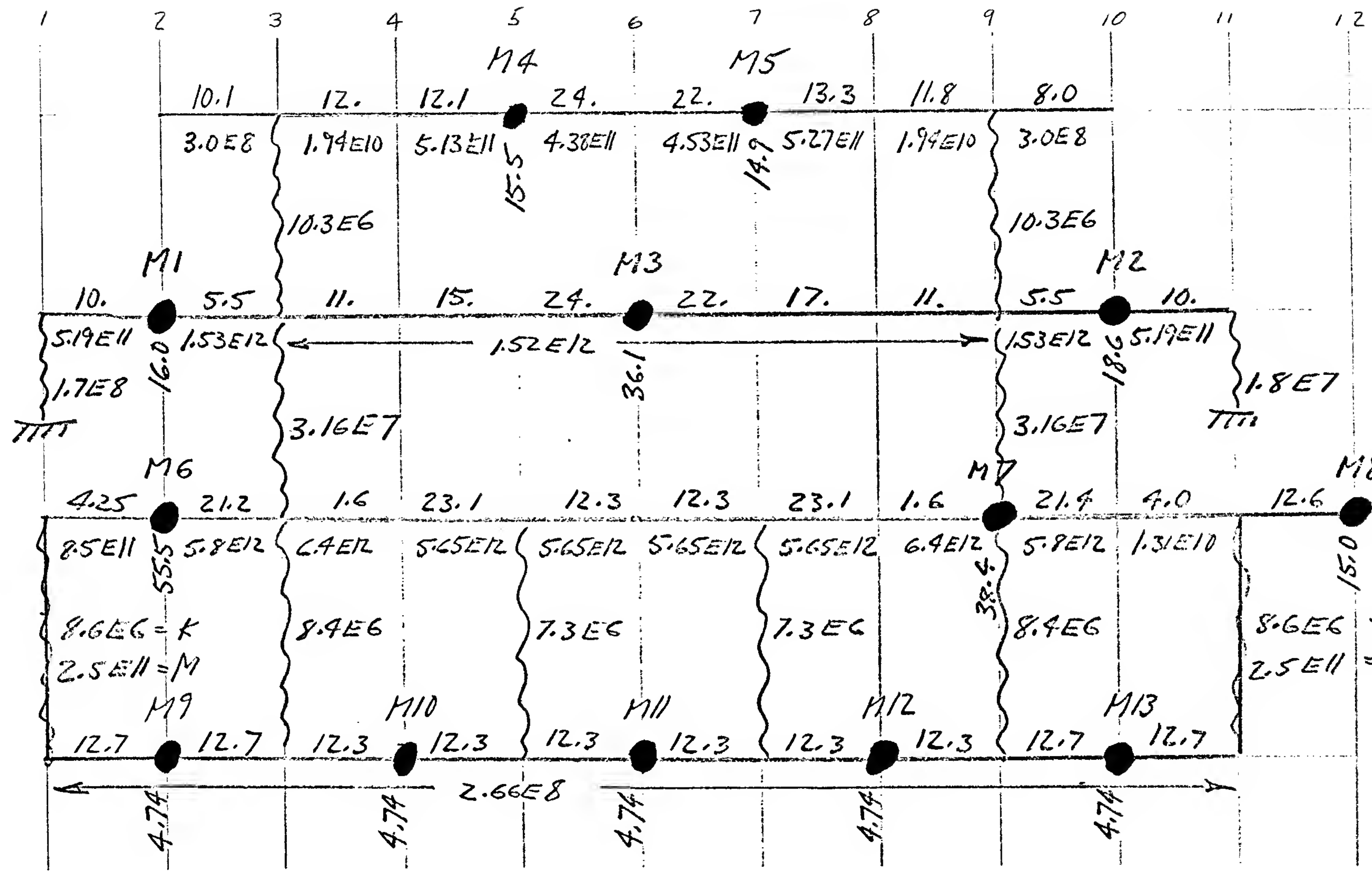

Exhibit 10: Jud Noble's Drawing of the LP Turbine Shock Model.

Mass - lb-sec²/in

Spring - lb/in

M= rotational restraint - in lb/rad

Beam $\frac{\text{Length - in}}{\text{EI - lb-in}^2}$